NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WARTIME REPORT

ORIGINALLY ISSUED
December 1941 as
Advance Restricted Report

INTERNALLY FINNED HONEYCOMB RADIATORS

By Arthur N. Tifford

Langley Memorial Aeronautical Laboratory
Langley Field, Va.

NACA

WASHINGTON

L - 492

INTERNALLY FINNED HONEYCOMB RADIATORS

By Arthur N. Tifford

L-492

SUMMARY

Calculations are made of the performance of several internally finned radiators and a comparison with the performance of conventional honeycomb radiators is made. If fins are placed inside conventional-size tubing, the hydraulic diameter of the air passages is reduced and the size and the power expenditure of the radiator are reduced. Calculations show that in a typical case the new radiator can be designed with 55 percent less volume and will require 18 percent less power expenditure than the conventional radiator dissipating the same amount of heat. If large tubes are internally finned in such a manner as to obtain the same hydraulic diameter for the air passages as is used today, the pressure drop and the power expenditure of radiators can be markedly reduced. For example, calculations show that the new radiator will require 20 percent less pressure drop and 18 percent less power expenditure than the conventional honeycomb radiator of the same volume dissipating the same amount of heat.

INTRODUCTION

The problem of designing the cooling installation in an airplane involves a compromise between the size and the power expenditure of the installation. Many ethylene-glycol radiators of various sizes and power costs will dissipate the amount of heat required by a given engine with a reasonable pressure drop. Small radiators tend to require large power expenditures; whereas large radiators tend to require small power expenditures. Because small radiators of low power expenditure are desirable, several methods of reducing the size of radiators without adversely affecting the power expenditure have been investigated in the past. Reducing the width of the liquid passage as much as possible has been found to be profitable because the same amount of cooling surface can thus be squeezed into a smaller volume. A limit on the width of the liquid passage is set by the clearance required by foreign materials in the coolant. Another method of increasing the amount of cooling surface per unit volume is to reduce the diameter of the radiator tubes. The large number of tubes required, however, brings on manufacturing difficulties. Present practice does not permit the use of radiator tubes of less than 0.20 inch diameter.

In this paper the performance characteristics of several radiators with internally finned tubes are calculated and are compared with the performance characteristics of conventional honeycomb radiators. The object of the paper is to present typical results to be expected from the use of internally finned radiators.

ANALYSIS OF THE PROBLEM

A direct method of reducing the size of radiators that is very promising is the placing of fins inside and along the complete length of conventional radiator tubing. Figure 1 shows various arrangements of fins in the tubing. Internal finning is an efficient means of obtaining small hydraulic diameters for the air passages of radiators without increasing the number of tubes that must be handled in the manufacturing processes. Internally finned radiators have, moreover, definite advantages over conventional honeycomb radiators with the same hydraulic diameter for the air passages.

Several factors explain the improved performance of internally finned radiators. In the conventional ethylene-glycol honeycomb radiator about 95 percent of the resistance to heat flow lies in the boundary layer on the air side. Internal finning makes the thermal resistances on the liquid and air sides more nearly equal because the surface exposed to the air is increased without a change in the surface on the liquid side. The reduction of the total resistance to the flow of heat for a given amount of surface causes a reduction in the amount of surface required for a given heat dissipation. One adverse effect occurs because indirect cooling surfaces, such as these internal fins, operate at lower average temperatures than the tube walls. Fin effectiveness of approximately 95 percent can be obtained, however, by using copper fins having width-to-thickness ratios of about 80.

Because the fins need be made only thick enough to efficiently carry the flow of heat (the fins do not have to supply structural strength to the radiator), because each fin contributes twice as much cooling-air surface per unit of metal weight as direct surface does, and because the

L-492

amount of coolant carried in the radiator is reduced, the weight of radiators can be reduced by internal finning. Furthermore, because of the fewer liquid passages and the thinner metal used, the ratio of open to total frontal area of an internally finned honeycomb radiator is greater than the ratio of open to total frontal area of a conventional honeycomb radiator with the same hydraulic diameter for the air passages. The greater open frontal area of an internally finned radiator allows more air passages to operate in parallel in a given frontal area. Consequently, the pressure drop required for cooling is less than the pressure drop required by a conventional radiator.

SYMBOLS

The following symbols, listed alphabetically, are used in this report. Consistent units are also given.

a quantity defined as $\sqrt{\dfrac{2h_a}{k_m t_f}}$

A total frontal area of radiator, square feet

A_o open frontal area of radiator, square feet

c_p specific heat at constant pressure, Btu per pound per °F

C_D drag coefficient of wing

C_L lift coefficient of wing

D hydraulic diameter of air passages, feet

D_t hydraulic diameter of radiator tubes, feet

E_f fin effectiveness

f free-area ratio, ratio of open frontal area to total frontal area

f_t free-area ratio, fins being neglected

f_1 friction factor $\dfrac{\Delta p_f D}{4qL}$

g acceleration of gravity, feet per second per second

h surface heat-transfer coefficient, Btu per second per square foot per $^{\circ}F$

$\frac{hD}{k}$ Nusselt number

h_a heat-transfer coefficient from air to tube wall, Btu per second per square foot per $^{\circ}F$

h_l heat-transfer coefficient from liquid to tube wall, Btu per second per square foot per $^{\circ}F$

H quantity of heat dissipated per unit time, Btu per second or horsepower

k thermal conductivity, Btu per second per square foot per $^{\circ}F$ per foot

k_1 thermal conductivity of air at radiator entrance, Btu per second per square foot per $^{\circ}F$ per foot

k_m thermal conductivity of metal, Btu per second per square foot per $^{\circ}F$ per foot

L radiator length, feet

n number of fins

P_D power required to force air through radiator, foot-pounds per second or horsepower.

P_W power required to support and propel weight of radiator, foot-pounds per second or horsepower

P_t total power required by radiator installation, foot-pounds per second or horsepower

p static pressure, pounds per square foot

Δp pressure drop across radiator, pounds per square foot

Δp_f pressure drop due to skin friction, pounds per square foot

q dynamic pressure, pounds per square foot

L-492

q_2 dynamic pressure at radiator exit, pounds per square foot

R Reynolds number $\left(\frac{\rho VD}{\mu}\right)$

R_a thermal resistance on air side $\left(\frac{1}{h_a S_a}\right)$

R_l thermal resistance on liquid side $\left(\frac{1}{h_l S_l}\right)$

s side of hexagonal tube, feet

S_a cooling surface, including fins, on air side, square feet

S_f cooling surface of fins, square feet

S_l cooling surface on liquid side, square feet

t tube-wall thickness, feet

t_f thickness of fin, feet

T absolute temperature, °F

T_0 absolute atmospheric temperature, °F

T_1 absolute temperature of air at radiator entrance, °F

T_2 absolute temperature of air at radiator exit, °F

T_l absolute liquid temperature, °F

T_w absolute wall temperature, °F

v open volume of radiator, neglecting finning, cubic feet

v_t total volume of radiator, cubic feet

V velocity, feet per second

V_1 velocity of air at radiator entrance, feet per second

V_0 air-stream velocity, feet per second

w width of minimum allowable passageway on liquid side, feet

w_f width of fin between tube walls, feet

W_f weight of fins, pounds

W_r weight of radiator without fins, pounds

α $\dfrac{\text{exit loss}}{2q_2}$

ϵ factor, approximately 1.5, by which to multiply radiator weight in order to account for additional required airplane structure

ρ air density, slugs per cubic foot

ρ_1 air density at radiator entrance, slugs per cubic foot

ρ_f weight density of fins, pounds per cubic foot of open volume of tubes

ρ_l coolant density, pounds per cubic foot

ρ_m metal weight density, pounds per cubic foot

ρ_r weight density of radiator without fins, pounds per cubic foot of open volume of tubes

η_p pumping efficiency of duct system

η_t heat-transfer efficiency

μ coefficient of viscosity, slugs per foot per second

μ_1 coefficient of viscosity of air at radiator entrance, slugs per foot per second

METHOD OF CALCULATION

The procedure followed in the calculations is similar to the procedure presented in reference 1 and has been adapted to take into account the effect of the thermal resistance on the liquid side.

The power expenditure chargeable to a wing or an engine–

L-492

nacelle cooling installation is the sum of the power required to push the cooling air through the radiator and duct system P_D/η_p and the power required to carry the radiator weight P_W:

$$P_t = \frac{P_D}{\eta_p} + P_W = \frac{A_0 V_1 \Delta p}{\eta_p} + \epsilon \frac{C_D}{C_L} V_0 (\rho_r + \rho_f) v \tag{1}$$

where, from reference 1,

$$\Delta p = \rho_1 V_1^2 \left[a\left(1 + \frac{T_w - T_1}{T_1}\eta_t\right) + \left(1 - \frac{f^2}{2}\right)\frac{T_w - T_1}{T_1}\eta_t + f_1 \frac{L}{D}\left(2 + \frac{T_w - T_1}{T_1}\eta_t\right)\right] \tag{2}$$

The heat dissipation is

$$H = g\rho_1 V_1 A_0 c_p (T_w - T_1)\eta_t \tag{3}$$

where

$$\eta_t = \frac{T_2 - T_1}{T_w - T_1} = 1 - e^{-\frac{h_a}{\rho_1 V_1 c_p g}\frac{4L}{D}}$$

or

$$\eta_t = 1 - e^{-4(0.024)\left(\frac{\rho_1 V_1 D}{\mu_1}\right)^{-0.2}\frac{L}{D}}$$

in the turbulent region.

The temperature difference $T_w - T_1$ is a function of the thermal resistance on the liquid side in internally finned radiators. The ratio of the thermal resistance on the two sides is

$$\frac{R_l}{R_a} = \frac{S_a h_a}{S_l h_l}$$

and, therefore, the available temperature difference with 100-percent effective fins is

$$T_w - T_1 = \frac{T_l - T_1}{1 + \frac{S_a h_a}{S_l h_l}}$$

Actually, the fin effectiveness is (reference 2)

$$E_f = \frac{\tanh \frac{aw_f}{2}}{\frac{aw_f}{2}}$$

where

$$a = \sqrt{\frac{2\,h_a}{k_m t_f}}$$

and w_f is the fin width between tube walls. The over-all effectiveness of the cooling surface on the air side is the sum of the fin surface multiplied by the fin effectiveness and the direct cooling surface divided by the total amount of surface on the air side $\left(\frac{S_f E_f + S_l}{S_a}\right)$.

The available temperature difference is, therefore,

$$T_w - T_1 = \frac{T_l - T_1}{1 + \frac{S_a h_a}{S_l h_l}} \frac{S_f E_f + S_l}{S_a} \qquad (4)$$

The detailed calculation of the performance of internally finned radiators is outlined in the appendix.

DISCUSSION

Typical results obtainable by the use of internally finned radiator tubes are represented in table I and figures 2, 3, and 4 by the values for radiators C, D, E, and F. Figure 2 shows the effect of increasing the relative thermal resistance on the liquid side. Each point corresponds to a radiator dissipating 1000 horsepower and having a frontal area of 4 square feet, an air passage length of 6 inches, and a hydraulic diameter of 1/12 inch. These radiators differ only in the amount of indirect cooling surface used on the air side. The number of fins is a measure of the amount of indirect cooling surface. The curves of figure 2 indicate that most of the gains derivable from internal finning without a change in the hydraulic diameter of the air passages are obtained by the use of three fins (radiator D, table I). In this case the power expenditure is reduced by 23 percent and the pressure drop is reduced by 40 percent. If radiator D were designed for the same pressure drop as radiator B (table I), the frontal area would be reduced 25 percent.

L-442

When small hydraulic diameters are used for the air passages of radiators, small volumes and small power expenditures are obtained. In practice, manufacturing difficulties associated with the large number of tubes required by the conventional honeycomb radiator restrict the minimum size of the tube diameter to about 0.20 inch. By the use of internal finning, however, small hydraulic diameters can be obtained for the air passages without increasing the number of tubes. The curves of figure 3 illustrate how internal finning reduces the size and power expenditure of the installation. Figure 4 shows that an increase in pressure drop is associated with a 50-percent reduction in radiator size. By a reduction in the radiator volume of 25 percent instead of 50 percent and by the use of a hydraulic diameter of 1/8 inch for the air passage, all three significant factors - the radiator volume, the power expenditure, and the pressure drop - would be reduced.

A comparison of the performances of radiators C and D (table I), which differ physically only in that the fins of radiator C are twice as thick as the fins of radiator D, emphasizes the fact that there exist limitations on the fin thickness other than weight alone. Radiator C, with the heavier fins, requires a higher pressure drop and a higher internal power expenditure than radiator D because the 1-percent increase in over-all cooling-surface effectiveness in radiator C, as compared with radiator D, does not nearly compensate for the 5-percent reduction in open frontal area caused by the doubly thick fins. The point to be emphasized is that the use of thicker fins does not necessarily mean that the required pressure drop and the power required to push the cooling air through the radiator will be reduced.

A large number of practical arrangements of internal fins exists. The hydraulic diameters for some of these are given in figure 1. Arrangements such as (c) or (f) in figure 1 are inherently better thermally than (b) or (e), respectively; that is, for a given weight of finning, a higher fin effectiveness is obtained. The hydraulic diameters are, however, more than 10 percent larger and the amounts of cooling surface are consequently 10 percent smaller. If all factors are considered, the over-all performance of installations finned like (c) as compared with (b), or like (f) as compared with (e), will differ negligibly when the air flow is turbulent.

As the hydraulic diameter of the air passages decreases with increased finning, the operating Reynolds number decreases. Thus, it becomes possible to operate in the transition range between laminar and turbulent flow or even at Reynolds numbers corresponding to completely laminar flow. In such cases small angles between fins are to be avoided because the laminar boundary layer will be extremely thick in the sharp corners and the surfaces forming the corners will accordingly have a greatly reduced effectiveness as far as friction and heat transfer are concerned. In effect, the fins will cause an increase in the radiator weight out of proportion to the increase in the heat dissipation per unit volume of radiator.

CONCLUSIONS

A large number of possible arrangements of internal fins for radiators exists. By the use of these internal fins, the power expenditure and the pressure drop required for cooling can be markedly reduced. In one typical case the new radiator is calculated to require 20 percent less pressure drop and 18 percent less power expenditure than the conventional honeycomb radiator of the same volume dissipating the same amount of heat.

Internal finning can also be used to reduce the hydraulic diameter of the air passages. This reduction will decrease the size and the power expenditure of radiators. Calculations show that in a typical case the radiator with internal finning requires 55 percent less volume and 18 percent less power expenditure than the conventional radiator having the same frontal area and dissipating the same amount of heat.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., October 17, 1941.

L-492

APPENDIX

CALCULATION OF THE PERFORMANCE OF INTERNALLY FINNED RADIATORS

The procedure followed in the calculation of the performance of internally finned radiators is similar to the procedure presented in reference 1.

For an assumed length and hydraulic diameter of the air passages, an approximate value for the operating Reynolds number is estimated. The corresponding value of Nusselt number is found from heat-transfer data, such as the data presented in figure 5. The heat-transfer coefficient h_a is determined and a first approximation to the fin effectiveness is obtained. The over-all effectiveness of the cooling surfaces having been determined, a first approximation to the available temperature difference $T_w - T_1$ is found from equation (4). The corresponding value of $\rho_1 V_1$ is found in figure 6, which has been plotted from equation (3) for the turbulent region. (In the transition and laminar regions, equation (3) must be used directly.) Reynolds number is now calculated and a second value for Nusselt number obtained, and the process is continued until the same value of $\rho_1 V_1$ is determined in two succeeding calculations. The pressure drop is now calculable from equation (2) and the data of figure 7; the power expenditure required is calculable from equation (1).

This procedure is repeated for each assumed radiator length. The power expenditure is then plotted against radiator volume as in figure 3.

The details of the sea-level calculations for the following assumed conditions are given in table I. Figures 3 and 4 present the results graphically.

The coolant is 97 percent ethylene-glycol solution and hexagonal tubes are used.

V_o = 734 fps (500 mph)

h_l = 0.20 Btu per sec per sq ft per °F

A = 4 sq ft

t = 0.005 in.

w = 0.028 in.

$c_p = 0.24$ Btu per lb per $^{\circ}F$

$H = 1000$ hp

$a = 0.1$

$T_o = 460^{\circ} + 73.5^{\circ} = 533.5^{\circ}$ F abs.

$T_l = 460^{\circ} + 290^{\circ} = 750^{\circ}$ F abs.

$\rho_l = 69.6$ lb per cu ft

$\rho_m = \rho_{copper} = 555$ lb per cu ft

$k_m = k_{copper} = 0.06055$ Btu per sec per sq ft per $^{\circ}F$ per ft

$\eta_p = 100$ percent

$\epsilon \dfrac{C_D}{C_L} = 0.1$

Allowance being made for complete adiabatic compression ahead of the radiator, the following quantities were calculated and used in the detailed calculations:

$T_1 = 460 + 118 = 578^{\circ}$ F abs.

$T_l - T_1 = 173^{\circ}$ F abs.

$k_1 = 4.063\,(10)^{-8}$ Btu per sec per sq ft per $^{\circ}F$ per ft

$\rho_1 = 0.00282$ slug per cu ft

$\mu_1 = 0.409$ slug per ft per sec

REFERENCES

1. Brevoort, M. J., and Leifer, M.: Radiator Design and Installation. NACA A.C.R., May 1939.

2. Harper, D. R., 3d, and Brown, W. B.: Mathematical Equations for Heat Conduction in the Fins of Air-Cooled Engines. Rep. No. 158, NACA, 1923.

L-492

TABLE I.- COMPUTED VALUES FOR SIX RADIATORS

$\frac{L}{D}$	ρ_1V_1 (slug/sq ft/sec)	R	η_t	$\frac{R_l}{R_a}$	$T_w - T_1$ (°F) (a)		$T_2 - T_1$ (°F)	$\frac{\Delta p}{\rho_1 V_1^2}$	Δp (lb/sq ft)	P_D (hp)	L (ft)	v (cu ft)	W_r (lb)	W_f (lb)	P_W (hp)	$P_D + P_W$ (hp)	v_t (cu ft)
Radiator A: $\frac{1}{4}$-in. hexagonal tubing, no fins, f = 0.75, A_o = 3.00 sq ft, ρ_r = 87 lb/cu ft																	
					100 percent												
40	0.481	24,600	0.387	0.0569	163.7		63.4	0.658	54.0	50.0	0.833	2.50	222	----	29.6	79.6	3.34
50	.390	19,900	.474	.0483	165.1		78.3	.835	45.1	34.0	1.041	3.13	276	----	36.8	70.8	4.16
60	.336	17,100	.547	.0428	165.9		90.7	1.031	41.2	26.8	1.250	3.75	330	----	44.2	71.0	5.00
70	.300	15,300	.611	.0390	166.5		101.7	1.219	38.7	22.4	1.458	4.37	386	----	51.4	73.8	5.84
Radiator B: $\frac{1}{12}$-in. hexagonal tubing, no fins, f = 0.470, A_o = 1.88 sq ft, ρ_r = 227 lb/cu ft																	
					100 percent												
60	0.475	8100	0.588	0.0675	162.0		95.3	1.185	94.8	54.6	0.417	0.78	181	----	24.2	78.8	1.56
70	.425	7200	.654	.0620	162.8		106.5	1.414	91.5	47.0	.486	.91	210	----	28.0	75.0	1.82
80	.398	6800	.695	.0571	163.6		113.7	1.622	91.1	43.8	.556	1.04	240	----	32.0	75.8	2.22
90	.377	6400	.730	.0529	164.4		120.0	1.835	92.5	42.2	.625	1.18	270	----	36.0	78.2	2.50
Radiator C: $\frac{1}{4}$-in. hexagonal tubing, fin arrangement, f = 0.678, A_o = 2.72 sq ft; average over-all effectiveness = 0.977; ρ_r = 87 lb/cu ft; t_f = 0.006 in.; D = $\frac{1}{12}$-in. = 0.0069 ft; S_a/S_l = 3; ρ_f = 53 lb/cu ft																	
					(percent) 100	97.7											
60	0.400	6800	0.587	0.170	147.8	144.4	84.7	1.180	66.8	46.2	0.417	1.25	113	67	24.0	70.2	1.67
70	.353	6000	.651	.154	150.0	146.6	95.4	1.395	61.7	38.0	.486	1.46	131	78	28.0	66.0	1.94
80	.328	5600	.693	.140	151.7	148.2	102.8	1.614	61.5	35.4	.556	1.67	149	89	31.8	67.2	2.22
90	.310	5300	.728	.130	153.2	149.7	109.0	1.840	62.7	34.4	.625	1.88	168	100	35.6	70.0	2.50

TABLE I.- COMPUTED VALUES FOR SIX RADIATORS

$\frac{L}{D}$	$\rho_1 V_1$ (slug/sq ft/sec)	R	η_t	$\frac{R_l}{R_a}$	$T_w - T_1$ ($^\circ$F) (a) (percent) 100	$T_w - T_1$ ($^\circ$F) (a) (percent)	$T_2 - T_1$ ($^\circ$F)	$\frac{\Delta p}{\rho_1 V_1^2}$	Δp (lb/sq ft)	P_D (hp)	L (ft)	v (cu ft)	W_r (lb)	W_f (lb)	P_W (hp)	$P_D + P_W$ (hp)	V_t (cu ft)
Radiator D: $\frac{1}{4}$-in. hexagonal tubing, fin arrangement, $f = 0.714$, $A_o = 2.86$ sq ft; average over-all effectiveness = 0.965; $\rho_r = 87$ lb/cu ft; $t_f = 0.003$ in.; $D = \frac{1}{12}$-in. = 0.0069 ft; $S_a/S_l = 3$; $\rho_f = 27$ lb/cu ft.																	
					100	96.5											
60	0.380	6400	0.588	0.162	148.8	143.6	84.4	1.176	60.2	42.2	0.417	1.25	113	33	19.6	61.8	1.66
70	.339	5800	.650	.147	150.8	145.5	94.6	1.359	55.4	34.6	.486	1.46	131	39	22.8	57.4	1.94
80	.317	5400	.686	.132	152.7	147.4	101.1	1.570	55.9	32.6	.556	1.67	149	44	26.0	58.6	2.22
90	.299	5300	.722	.123	154.2	148.8	107.4	1.772	56.0	30.8	.625	1.88	168	50	29.2	60.0	2.50
Radiator E: $\frac{3}{4}$-in. hexagonal tubing, fin arrangement, $f = 0.848$, $A_o = 3.40$ sq ft; average over-all effectiveness = 0.900; $\rho_r = 42$ lb/cu ft; $t_f = 0.012$ in.; $D = \frac{1}{4}$-in. = 0.0208 ft; $S_a/S_l = 3$; $f_t = 0.904$; $\rho_f = 18$ lb/cu ft																	
					100	90.0											
40	0.422	21,500	0.499	0.212	142.9	128.6	64.2	0.664	41.8	38.4	0.83	3.01	131	54	24.8	63.2	3.34
50	.345	17,600	.594	.180	146.8	132.1	78.4	.841	35.5	26.8	1.04	3.77	163	67	30.8	57.6	4.16
60	.299	15,300	.670	.160	149.2	134.3	90.0	1.030	32.8	21.6	1.25	4.52	194	80	36.6	58.2	5.00
70	.272	13,900	.733	.149	150.7	135.6	99.5	1.231	32.2	19.2	1.46	5.27	226	94	42.6	61.8	5.84
Radiator F: $\frac{3}{8}$-in. hexagonal tubing, fin arrangement, $f = 0.751$, $A_o = 3.00$ sq ft; average over-all effectiveness = 0.936; $\rho_r = 65$ lb/cu ft; $t_f = 0.0039$ and 0.0045 in.; $D = 0.0066$ ft; $S_a/S_l = 4.73$; $f_t = 0.820$; $\rho_f = 47$ lb/cu ft																	
					100	93.6											
60	0.370	6000	0.647	0.273	135.9	127.2	82.3	1.187	58.0	41.9	0.396	1.30	84	59	19.2	61.1	1.58
70	.336	5400	.709	.255	137.8	129.0	91.5	1.407	56.7	37.2	.462	1.52	99	69	22.4	59.6	1.85
80	.305	4900	.763	.237	139.9	131.0	100.0	1.633	54.2	32.3	.528	1.73	113	79	25.6	57.9	2.11
90	.286	4600	.806	.225	141.2	132.2	106.6	1.865	54.4	30.4	.594	1.95	127	89	28.8	59.2	2.38

[a] The temperature difference $T_w - T_1$ has been given for $E_f = 100$ percent and also for average values of E_f when different from 100 percent.

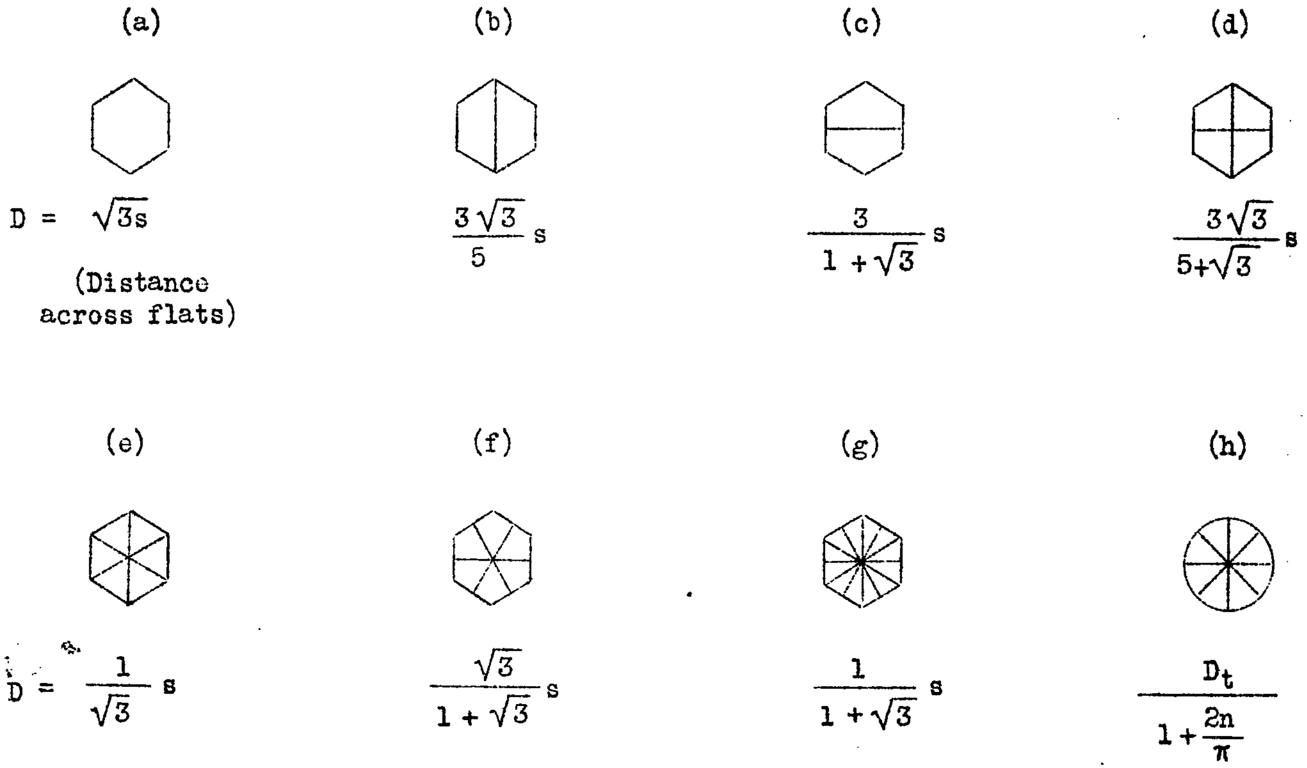

Figure 1.- Several arrangements of internal cooling surfaces.

L-492

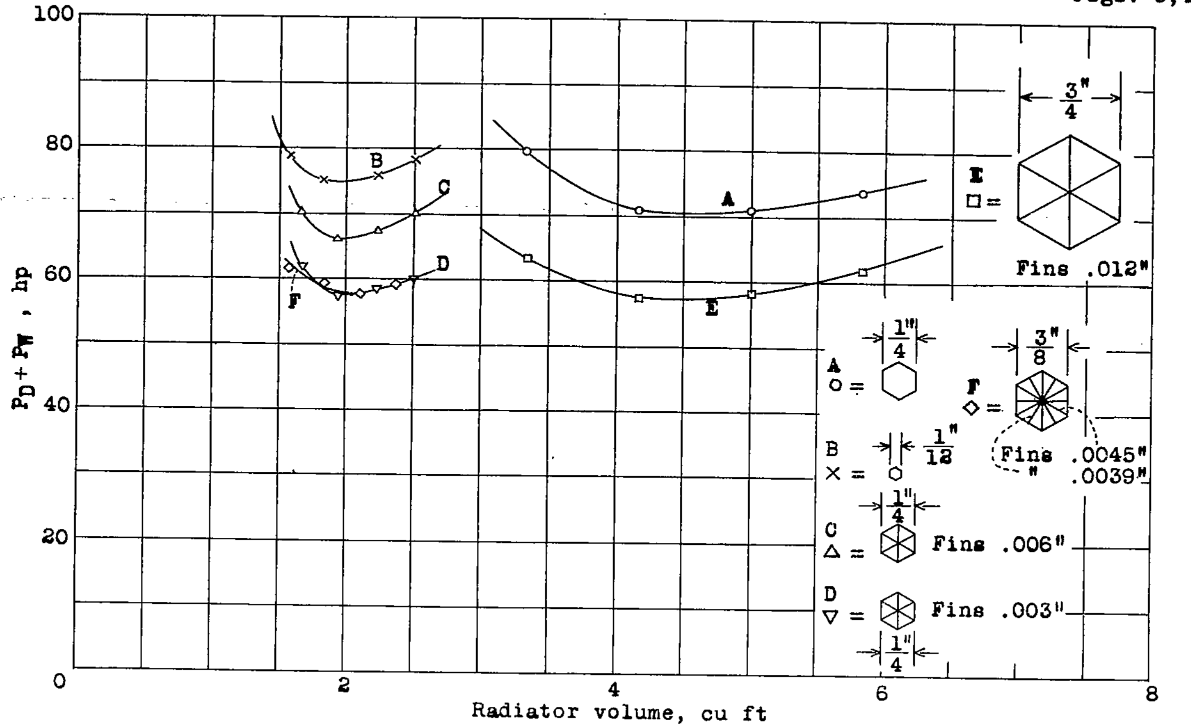

Figure 3.- Power expenditure of radiators dissipating 1000 horsepower.

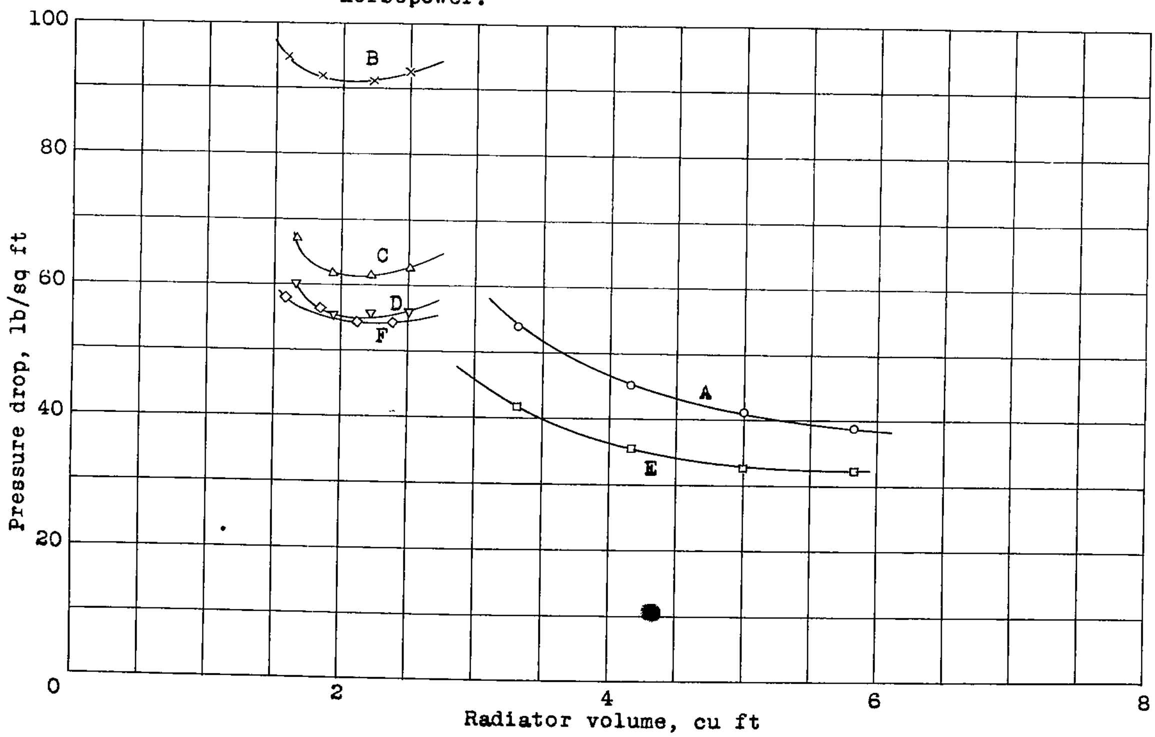

Figure 4.- Pressure drop across radiators dissipating 1000 horsepower.

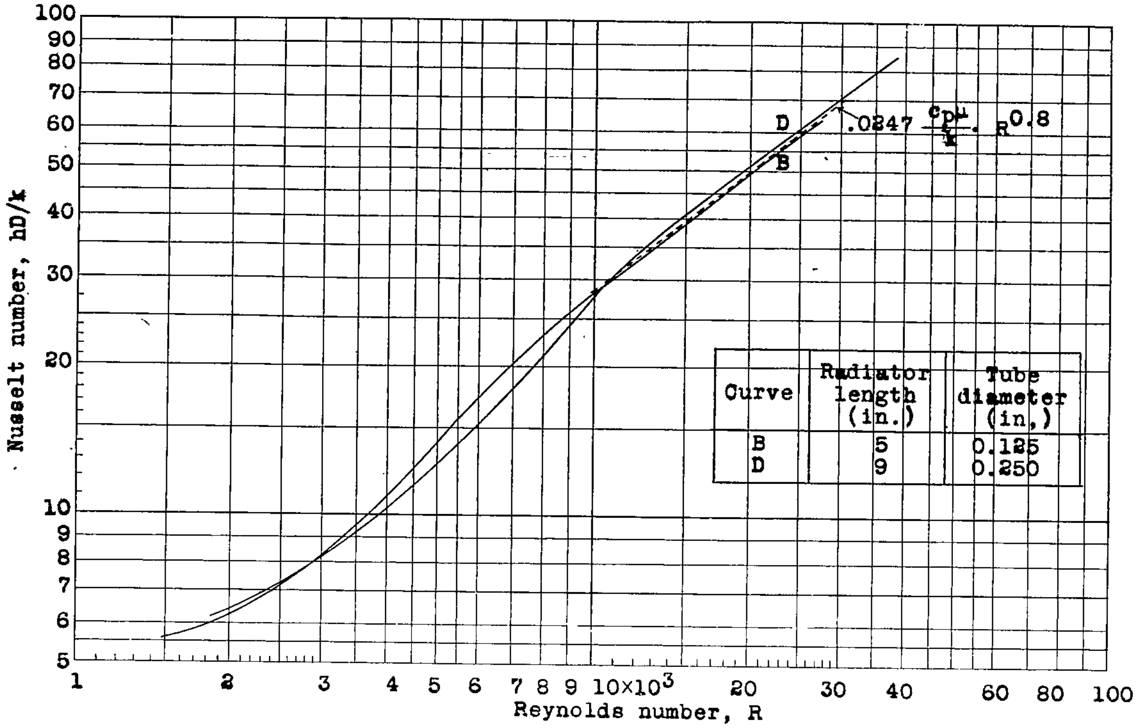

Figure 5.- Heat-transfer data taken from reference 2.

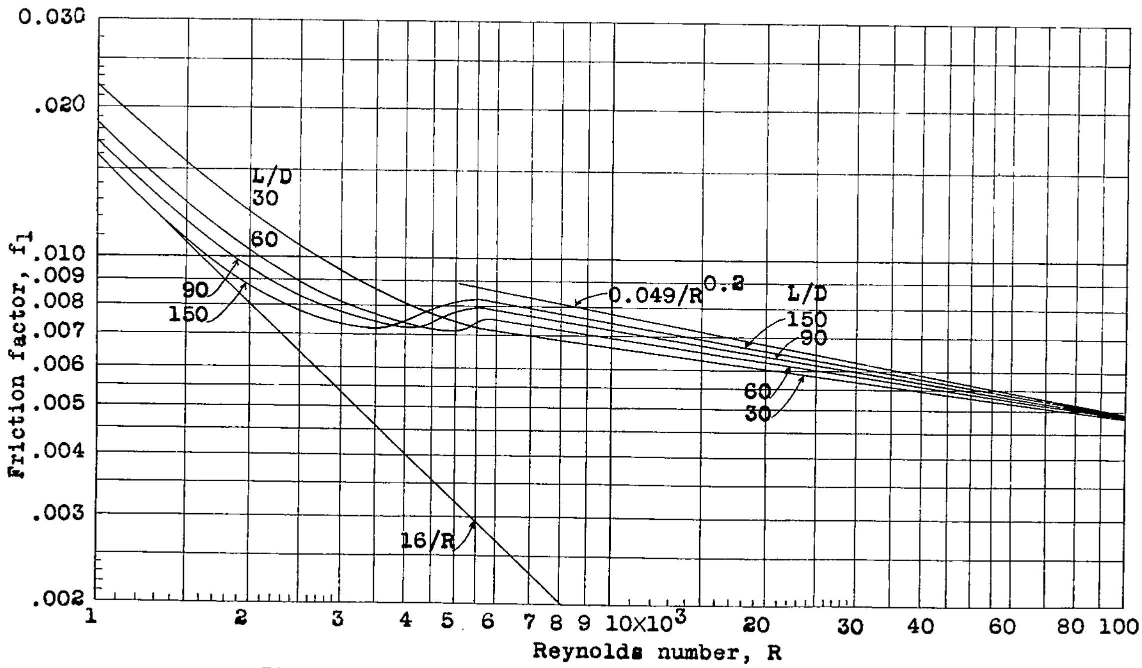

Figure 7.- Friction data taken from reference 2.

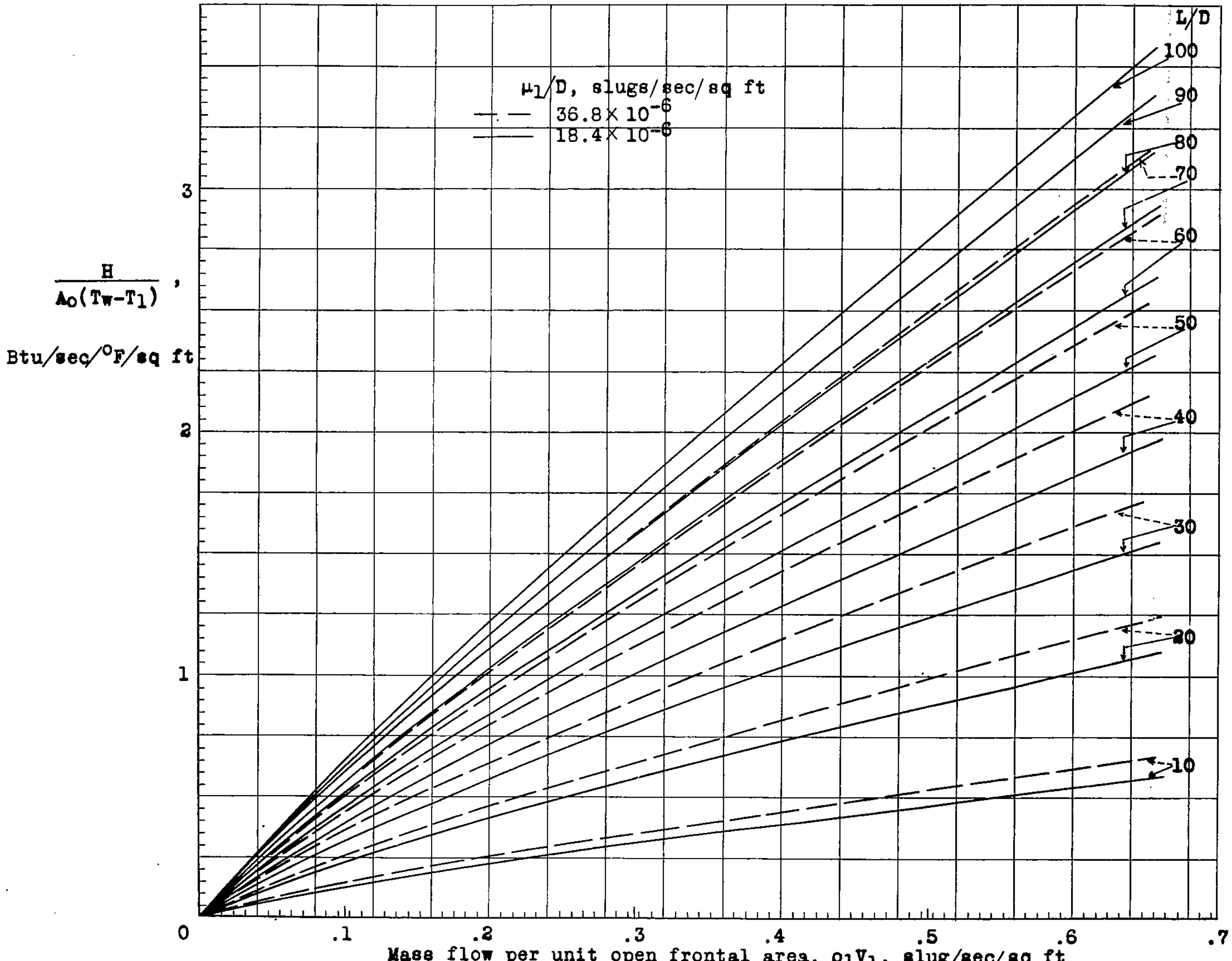

Figure 6.- The ratio $H/A_o(T_w-T_1)$ as a function of $\rho_1 V_1$ with L/D and μ_1/D as parameters.